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Lubricant Effects on Efficiency of a Helicopter Transmission

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LUBRICANT EFFECTS ON EFFICIENCY OF A HELICOPTER TRANSMISSION

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SUMMARY

Efficiency tests were conducted using eleven different lubricants in the NASA Lewis Research Center's 500 hp torque regenerative helicopter transmission test stand. The test transmission was the OH58A helicopter main transmission. The mechanical power input to the test transmission was 224kW (300 hp) at 6060 rpm. Tests were run at oil-in temperatures of 355°K (180°F) and 372°K (210°F). The efficiency was calculated from a heat balance on the water running through an oil-to-water heat exchanger while the transmission was heavily insulated.

The following results were obtained.

1. Among the eleven different lubricants, the efficiency ranged from 98.3 to 98.8 percent, which is a 50 percent variation relative to the losses associated with the maximum efficiency measured.

2. For a given lubricant, the efficiency increased as temperature increased and thus as viscosity decreased. There were two exceptions which could not be explained on the basis of available data.

3. Between lubricants, efficiency was not correlated with viscosity. There were relatively large variations in efficiency with the different lubricants whose viscosity generally fell in the 5 to 7 centistoke range.

4. The lubricants had no significant effect on the vibration signature of the transmission.

INTRODUCTION

The mechanical efficiency of helicopter power train components is generally very high. As a rule of thumb, there is a loss of 3/4 percent for a planetary stage, and 1/2 percent for a single gear mesh. More specific estimates may be found in reference 1. An important step in development of the power transmission path in helicopters is to do everything possible to minimize power losses. Minimizing the power loss makes it possible to extend the performance envelope for the helicopter. Range, payload, and operating ceiling can be increased if efficiency is increased. With large, high power helicopter applications only a few tenths of one percent mechanical power loss can be the equivalent to the loss of hundreds of kilowatts. Compared with total power used this loss may seem trivial from an energy conservation viewpoint, but the effect on the operating envelope may be more significant. Since all mechanical power losses must be dissipated as heat, improvements in transmission efficiency will permit smaller and lighter weight cooling systems. This effect adds to increase the payload capacity of the helicopter.

The total power loss in a helicopter transmission is a function of many parameters. Sliding friction losses in the gears, bearings, and seals contribute a large effect. Sliding losses occur in whatever lubrication regime is present, whether the regime is hydrodynamic, elastohydrodynamic, boundary lubrication, or some mixture of these. Other large contributors to the losses are windage losses and lubricant churning losses in the rotating components. To a lesser extent rolling traction losses and material hysteretic losses are also contributors to total power loss. In a high speed transmission it is expected that a variety of physical and chemical characteristics of the oil influence the operating efficiency.

Martin (ref 2) presented a comprehensive review and bibliography of power loss calculations for friction between gear teeth. Martin (ref 3) concentrates on the problem of calculating the losses in the tooth contact. Anderson and Loewenthal (ref 4) give a more encompassing method of estimation of power losses which extends the calculation to partially loaded gear sets, including bearings. Bearing power loss was earlier addressed by Townsend, Allen, and Zaretsky (ref 5). Martin (refs 2 and 3) has pointed out that efficiency is important since it directly affects the cooling requirements of the gears. Townsend and Akin (refs 6 to 8) have studied gear tooth cooling and concluded that for best efficiency and cooling, the gears should be jet lubricated with radially directed jets on the exit side of the gear mesh.

Murphy, et al (ref 9) have studied the effect of lubricant traction on worm gear efficiency. They found that synthetic oils with lowest traction coefficients gave the best efficiency. This is to be expected since traction losses are the largest component of total loss in low speed worm gear sets which normally do not have much churning and windage losses.

In view of the above, the objective of the work presented herein was to measure the operating efficiency of a helicopter transmission with eleven different commercially available lubricants. A further objective was to examine the measured results for correlation with available physical property data on the lubricants and thereby determine reasons for the variability in efficiency from one lubricant to another.

APPARATUS, SPECIMENS, AND PROCEDURE

Transmission Test Stand

Figure 1 shows the NASA 500 HP helicopter transmission test stand, which was used to run the efficiency tests. The test stand operates on the "four-square" or torque regenerative principle, where mechanical power is recirculated around the closed loop of gears and shafting, passing through the test transmission. A 149kW (200hp) SCR controlled DC motor is used to power the test stand and control the speed. Since the torque and power is recirculated around the loop, only the losses due to friction have to be replenished.

A 11kW (15hp) SCR controlled DC motor driving against a magnetic particle clutch is used to set the torque in the test stand. The output of the clutch does not turn continuously, but only exerts a torque through the speed reducer gearbox and chain drive to the large sprocket on the differential gear unit. The large sprocket is the first input to the differential. The second input is from the upper shaft which passes concentrically through the hollow upper gear shaft in the closing end gearbox. The output shaft from the differential gear unit is the previously mentioned hollow upper gear shaft of the closing end gearbox. The torque in the loop is adjusted by changing the electrical field strength at the magnetic particle clutch. The 11kW (15hp) motor was set to turn continuously at 70 rpm.

The input and output shafts to the test transmission are equipped with speed sensors, torque meters, and slip rings.

Figure 2 is a schematic of the efficiency measurement system. The system allows the helicopter transmission to be operated in a thermally insulated environment with provisions to collect and measure the heat generation due to mechanical power losses in the transmission. In this schematic, the instrumentation used to measure torque and speed, and hence power input to the test transmission is not shown. The original oil-to-air heat exchanger which is standard flight hardware was replaced with an oil-to-water heat exchanger so as to allow more precise measurements of the heat rejection during an efficiency test run. By using the water to remove heat, any uncertainty of the correct value for specific heat of the oil was removed.

Figure 3 shows the test transmission mounted in the test stand. Figure 4 shows the test stand with the insulated housing around the test transmission. Thermocouples were placed at various locations inside the insulated housing to verify the adequacy of the insulation.

Test Lubricants

Tables 1 to 4 describe the lubricants used, their specification, physical properties and generic identification. All the lubricants were tested for physical properties, contaminents, and wear particles prior to and after completion of all test runs, as further described herein. Table 5 lists supplemental data related to the lubricants in this study which was gathered from references 10-12. All the lubricants were near to the 5-7 centistoke range in viscosity and were qualified for use or considered likely candidates for use in helicopter transmissions. Lubricants A and B are automatic transmission fluids (ref 13).

Test Transmission

The test transmission was the main rotor transmission from the U.S. Army's light observation helicopter (OH-58) as described in reference 14 and shown in figure 5. The transmission is rated for 201kW (270hp) continuous duty and 236kW (317 horsepower) at takeoff for 5 minutes. The 100 percent input speed is 354 rpm. The input shaft drives a 19 tooth spiral bevel pinion. The pinion meshes with a 71 tooth gear. The input pinion shaft is mounted on triplex ball bearings and one roller bearing. The 71 tooth bevel gear is carried on a shaft mounted in duplex ball bearings and one roller bearing. The bevel gear shaft drives a floating sun-gear which has 27 teeth. The power is taken out through the planet carrier. There are three planet gears of 35 teeth which are mounted on spherical roller bearings. The ring gear (99 teeth) is splined to the top case and therefore is stationary. The overall gear ratio is 17.44:1 reduction.

The planet bearing inner races and rollers are made of AISI M-50 steel. The outer races and planet gears, which are integral, are made of AISI 9310. The cage material is 2024-T4 aluminum. The gear shaft duplex bearing material is CVM 52CB. All other bearings are made of AISI 52100 with bronze cages. The sun gear and ring gear material is Nitralloy N (AMS6475). The input spiral bevel gear-set material is AISI 9310. Lubrication is supplied through jets located in the top case.

Test Procedure

Before the start of each efficiency test, the transmission and heat exchanger were cleaned out with solvent and the transmission components were visually inspected. Gear tooth surfaces were photographed. The transmission was then assembled and mounted in the test stand and filled with oil. The rig was run briefly to check for oil leaks. Then the loose fill insulation was added, filling the plexiglass box to completely surround and thermally insulate the test apparatus and transmission.

Efficiency test runs were made with the oil inlet temperature controlled to within less than one degree kelvin. Tests were run at oil inlet temperatures of approximately 355°K (180°F) and 372°K (210°F). The torque on the input shaft was 352 N-m (3118 lb-in) for each run. The input speed was 6060 rpm. This

corresponds to the full power condition on the test transmission. The oil inlet and oil outlet temperatures were monitored until equilibrium conditions were established, which generally took about 20-30 minutes. Then the efficiency test run was started. Water was collected in the weighing tank and data was recorded for total water weight, inlet and outlet temperatures for the water and oil, and flow rate for the water and oil. Vibration spectrum records were made for seven accelerometers mounted on the test transmission. Data logging records were taken once each minute for a total test time of approximately 30 minutes for each test temperature.

After the tests were completed the transmission was disassembled, cleaned and visually inspected for changes in the gear and bearing surfaces. Photographic records were made. The lubricant was saved for later analysis. The efficiency was later calculated from the heat balance on the water that flowed through the heat exchanger.

RESULTS AND DISCUSSION

The experimentally determined efficiencies are listed in table 6 and plotted against oil inlet temperature in figure 6. The range of efficiencies varied from 98.3 to 98.8 percent. This is an overall variation in losses of almost 50 percent, relative to the losses associated with the maximum efficiency measured.

In general, the higher test temperature for a given lubricant yielded a higher efficiency. The exceptions were with lubricants E and C, which were different types of synthetic lubricant. Lubricant G, being more viscous than the other lubricants could not be tested at the targeted oil inlet temperature. This was because the heat generated could not be removed with the existing water/oil heat exchanger. The test temperature floated up to 378.5°K with the heat exchanger at full water flow capacity. At the higher temperature the efficiency for oil G was consistent with the efficiencies lower viscosity oils. The two automatic transmission fluids (A and B) and the Type I Synthetic Gear Lubricant (E) yielded significantly lower efficiencies as a group.

In figure 7 the efficiencies are plotted against the lubricant viscosity at the inlet temperature. This was done to determine if the efficiency is strongly dependent on the viscosity. By the plotted results, it is clear that viscosity variation is not the primary reason for the varying efficiencies between the different lubricants. But there is a general trend to higher efficiency for lower viscosity for all the lubricants except C and E. The slope of the aforementioned trend is identical for a large number of the lubricants.

The reason for the lower efficiency for lubricants A, B, and E is suspected to be related to higher traction coefficient characteristics, which would come into effect in the elastohydrodynamic regime of lubrication between the gear teeth. It is interesting to note that while the Mil-L-7808 lubricant was the lowest viscosity oil, the efficiency was no better than the Mil-L-23699 lubricants. This may also be related to an EHD tractional or frictional phenomenon. The reason for the reverse trends with viscosity for lubricants E and C is unknown at this time.

The vibration spectra were monitored during the tests with the various lubricants. The variations in amplitude were insignificant from one oil to the next. Figure 8 is an typical vibration spectrum measured by placing an accelerometer on the transmission case at the split line between the top and bottom cases.

Tables 7-10 give the comparison between the lubricant analyses performed before and after the efficiency test runs. It is noticed that lubricants A and C showed significant increases in the iron content (table 7). Also, lubricant E showed a strong acid value before and after the test runs (table 8). These three lubricants were among the ones giving deviant performances for efficiency.

The visual inspection of the transmission components after each test run showed no indications of wear or degradation. In fact, the black oxide coating which was placed on the gear surfaces during manufacturing was hardly worn off.

SUMMARY AND RESULTS

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Table 1
Test lubricant types

CODE NO.	SPECIFICATION	TYPE
A	DEXRON II GM 6137-M	Automatic Transmission Fluid
B	DEXRON II GM 6137-M	Automatic Transmission Fluid
C	MIL-L-23699	Turbine Engine Oil
D	MIL-L-23699	Type II Synthetic Gas Turbine Engine Oil
E		Type I Synthetic Gear Lubricant
F		Synthetic Paraffinic with Antiwear Additives
G	MIL-L-2104C MIL-L-46152	Synthetic Fleet Engine Oil
H	MIL-L-7808	Turbine Engine Oil
I	MIL-L-23699	Type II Turbine Engine Oil
J	MIL-L-23699	Type II Turbine Engine Oil
K		Turbine Engine Oil

Table 2
Specific Gravity Data According to ANSI/ASTM Specification D-1481,
API Gravity According to ANSI/ASTM *Specification D-1298

LUBRICANT	SPECIFIC GRAVITY @ LISTED TEMP			API GRAVITY 288°K
	313°K	355°K	373°K	
A	.8620	.8558	.8514	29.8
B	.8626	.8548	.8546	29.9
C	.9973	.9862	.9843	8.2
D	.9868	.9768	.9746	9.7
E	.9322	.9211	.9201	17.7
F	.8262	.8108	.8088	36.0
G	.8629	.8536	.8527	29.6
H	.9442	.9320	.9313	15.7
I	.9659	.9568	.9546	12.8
J	.9856	.9759	.9747	10.1
K	.9829	.9721	.9725	10.3

*ANSI/ASTM, American National Standards Institute/American Society for Testing and Materials

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Table 3
Kinematic viscosity data according to
ANSI/ASTM Specification D-455

LUBRICANT	VISCOSITY @ LISTED TEMP, CSt		
	313°K	355°K	373°K
A	37.48	10.48	7.01
B	33.15	9.64	6.52
C	26.40	7.69	5.12
D	26.17	7.50	5.00
E	33.91	8.91	5.87
F	28.01	8.15	5.36
G	56.65	15.05	9.83
H	13.16	4.73	3.38
I	24.19	7.18	4.85
J	24.76	7.23	4.89
K	26.39	7.61	5.09

Table 4
Specific heat data according to
ANSI/ASTM Specification D-3947-80

LUBRICANT	SPECIFIC HEAT @ LISTED TEMPERATURE					
	313°K		373°K		413°K	
	Cp	σ	Cp	σ	Cp	σ
A	.42	.091	.42	.12	.44	.14
B	-	-	-	-	-	-
C	.33	.097	.32	.097	.32	.091
D	.33	.071	.34	.072	.34	.084*
E	.68	.11	.73	.13	.76	.20
F	.53	.12	.54	.13	.54	.14
G	.50	.091	.47	.058	.42	.059
H	.37	.0[illegible]6	.30	.037	.31	.094
I	.53	.060	.47	.039	.44	.075*
J	-	-	-	-	-	-
K	.44	.073	.38	.076	.34	.075

*For calculation of Cp and σ (std. deviation) one value, inordinately different from the others, was discarded. Thus, four values rather than five were used to determine these data.

Table 5
Pressure-viscosity coefficients for test lubricants expressed as reciprocal asymtotic isoviscous pressures

LUBRICANT	RECIPROCAL ASYMTOTIC ISOVISCOUS PRESSURE, α^*, $(N/m^2)^{-1}$ @ LISTED TEMPERATURE			SOURCE OF DATA
	311°K	372°K	422°K	
A	1.35×10^{-8}	$.951 \times 10^{-8}$	$.772 \times 10^{-8}$	ref 11
B	a	a	a	
C	-	1.01×10^{-8}	$.832 \times 10^{-8}$	ref 10
D	b	b	b	
E	-	-	-	
F	1.90×10^{-8}	1.50×10^{-8}	1.15×10^{-8}	ref 12[c]
G	1.42×10^{-8}	1.02×10^{-8}	$.918 \times 10^{-8}$	ref 11
H	-	$.894 \times 10^{-8}$	$.731 \times 10^{-8}$	ref 10
I	b	b	b	
J	b	b	b	
K	1.28×10^{-8}	$.987 \times 10^{-8}$	$.851 \times 10^{-8}$	ref 12

[a] most likely the same as A since they are similar lubricants
[b] most likely the same as C or K since they are similar lubricants
[c] estimate based on ref 12

Table 6
Measured efficiencies

LUBRICANT	EFFICIENCY	INLET TEMP, °K
A	.9840 .9850	361.5 375.0
B	.9833 .9843	356.8 375.0
C	.9876 .9873	356.4 371.5
D	.9860 .9874	356.1 370.1
E	.9835 .9832	361.0 371.5
F	.9865 .9877	355.7 372.0
G	.9873	378.7
H	.9870 .9879	355.6 372.1
I	.9864 .9882	355.6 372.2
J	.9864 .9877	355.6 372.3
K	.9869 .9882	355.6 372.3

Table 7
Total iron analysis by
calorimetric method (ref 15)

LUBRICANT	IRON CONTENT (ppm)	
	BEFORE TEST	AFTER TEST
A	1	4
B	< 1	< 1
C	1	6
D	< 1	1
E	< 1	1
F	< 1	2
G	2	3
H	< 1	1
I	< 1	< 1
J	< 1	< 1
K	< 1	< 1

Table 8
Lubricant acid analysis according to
ANSI/ASTM Specification D-664

LUBRICANT	TOTAL ACID NUMBER Mg KOH/g	
	BEFORE TEST	AFTER TEST
A	.54	.54
B	-	-
C	.01	.02
D	.07	.07
E	15.8*	15.7†
F	.42	.51
G	3.2	3.5
H	.34	.34
I	.34	.38
J	-	-
K	.48	.43

* Strong acid value = 7.1 on sample

† 6.2 acid value

Table 9
Particulate contamination count according to
SAE Aerospace Recommended Practice ARP 598A

LUBRICANT BEFORE AFTER	Number of Particles/100ml Particle Sizes in Micrometers					
	5-15	15-25	25-50	50-100	100	Fibers
A	17 4	2 1	2 6	4 7	10 11	12 10
B	-	-	-	-	-	-
C	72 4	36 1	18 2	12 1	10 5	7 9
D	685 200	275 65	35 38	22 24	15 21	20 39
E	120 44	60 7	23 10	25 13	22 12	33 19
F	60 475	16 8	30 2	13 5	7 6	22 52
G	49 4740	39 10	45 11	38 9	34 6	78 34
H	1780 1850	72 118	45 108	40 60	25 52	32 62
I	54 840	23 660	17 450	16 210	4 80	19 120
J	-	-	-	-	-	-
K	185 105	175 48	100 35	70 21	35 20	45 22

Table 10
Wear metals test results using x-ray fluorescence filter method (ref 16)

LUBRICANT BEFORE AFTER	ELEMENTS (PPM)												LIMIT(3) OF DETECTION (PPM)
	Mg	Al	Cl	Fe	Ni	Cu	Pb	Zn(1)	P(2)	S(2)	Ca(2)	Ba(2)	
A	0.48 -	- 5.91	2.47 1.12	- 0.51	- 0.10	- 0.14	0.21 -	- 0.11	0.18 0.17	4.71 1.12	- -	0.23 0.12	0.11 0.09
B	-	-	-	-	-	-	-	-	-	-	-	-	-
C	0.28 -	- 2.97	0.73 1.04	0.13 2.19	- 0.21	- 0.12	- -	- 0.15	[illegible] 0.19	- 0.20	- -	- -	0.09 0.09
D	0.27 -	- 12.7	0.90 2.08	- 1.16	- 0.24	- 0.19	- 0.20	- 0.20	0.16 0.71	- 0.51	- -	- -	0.11 0.15
E	0.16 0.12	0.19 1.69	7.57 1.61	0.10 0.26	- -	- 0.11	1.28 -	7.27 3.71	2.15 0.94	13.01 4.29	[illegible] -	10.16 2.43	0.09 0.09
F	0.31 5.36	- -	0.45 2.49	- -	- -	- -	- -	- -	0.19 2.42	7.08 51.0	- -	- -	0.10 0.55
G	1.31 0.39	- 0.67	4.91 1.49	- 0.22	- -	- -	- -	1.51 0.39	0.70 -	5.29 0.89	8.69 2.53	- -	0.43 0.13
H	0.29 0.67	- 4.68	3.81 16.68	0.11 0.74	- -	- 0.26	0.16 -	- 0.62	0.47 2.37	0.21 3.20	- 3.47	- -	0.10 0.25
I	0.33 0.34	- 1.18	0.56 0.85	- 0.58	- -	- -	0.11 0.12	- 0.13	0.58 0.44	- 0.16	- -	- -	0.10 0.11
J	-	-	-	-	-	-	-	-	-	-	-	-	-
K	0.60 1.26	- 0.39	9.80 7.30	0.28 0.56	- -	- -	- 0.65	- -	2.51 1.86	- -	- -	- -	0.24 0.37

(1) Zn could be due to wear when present with copper, or as an additive when present alone.
(2) P, S, Ca, Ba probably present as additives.
(3) Limit of detection for sample, when - shown, element is less than this value.

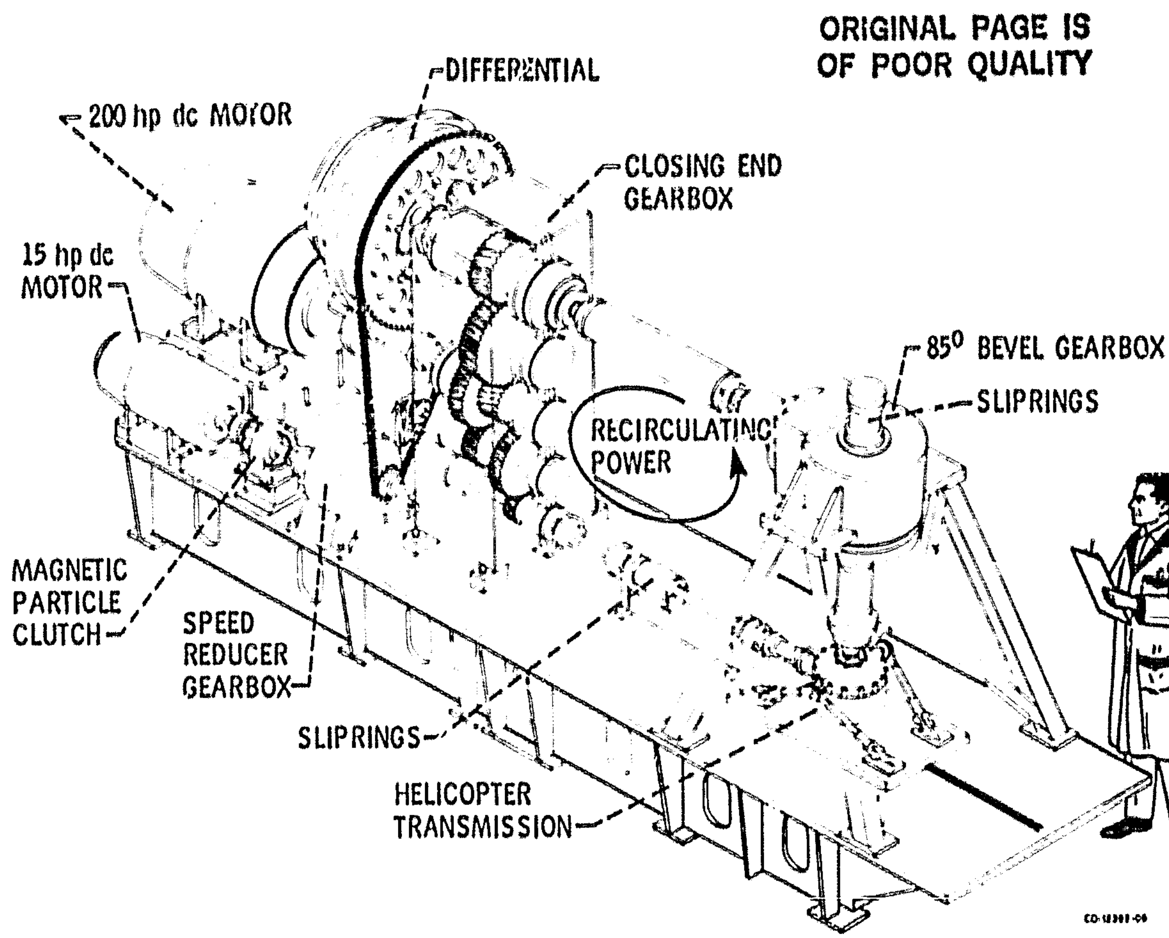

Figure 1. - NASA 500 hp helicopter transmission test stand.

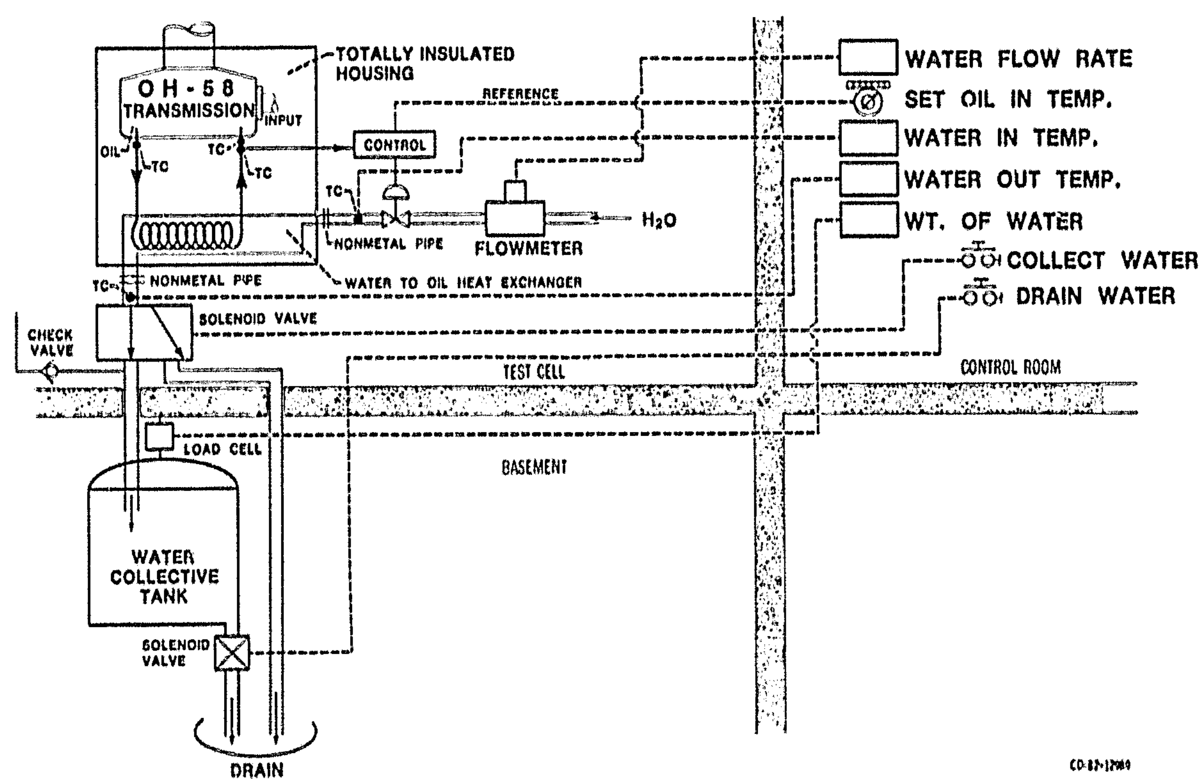

Figure 2. - Schematic of measurement system.

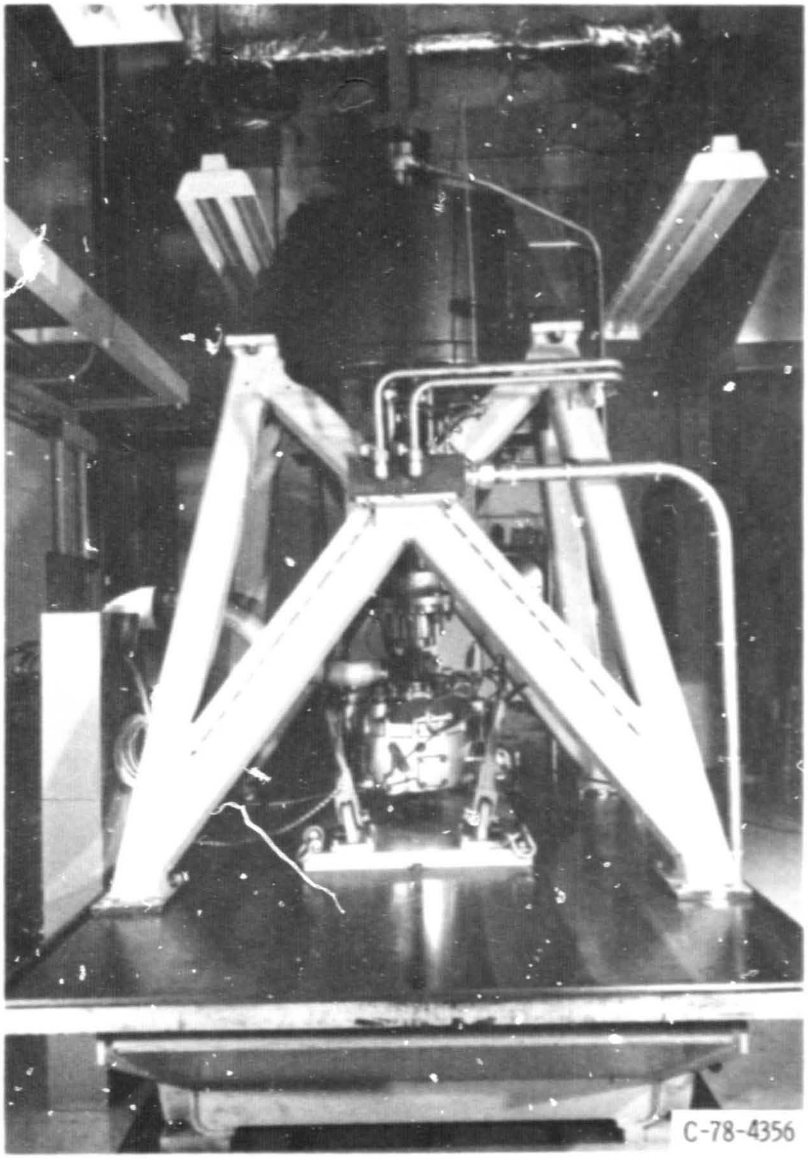

Figure 3. - View of test stand showing OH-58 transmission installed.

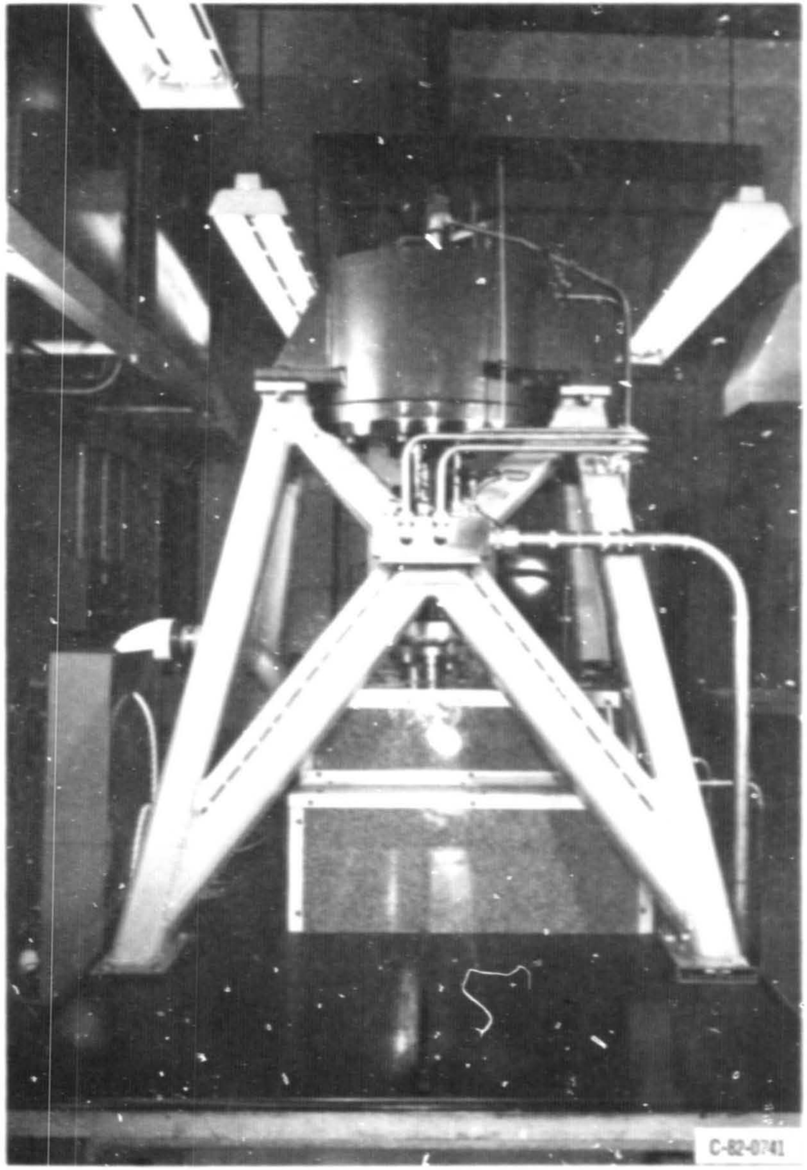

Figure 4. - View of test stand showing insulated transmission housing.

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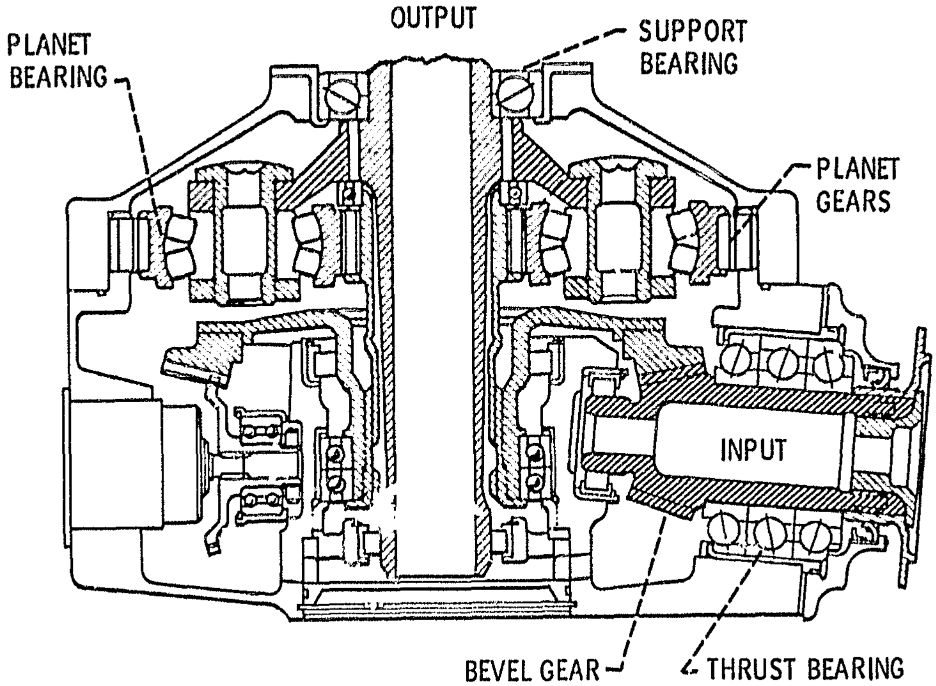

Figure 5. - Cross section of OH-58 helicopter transmission.

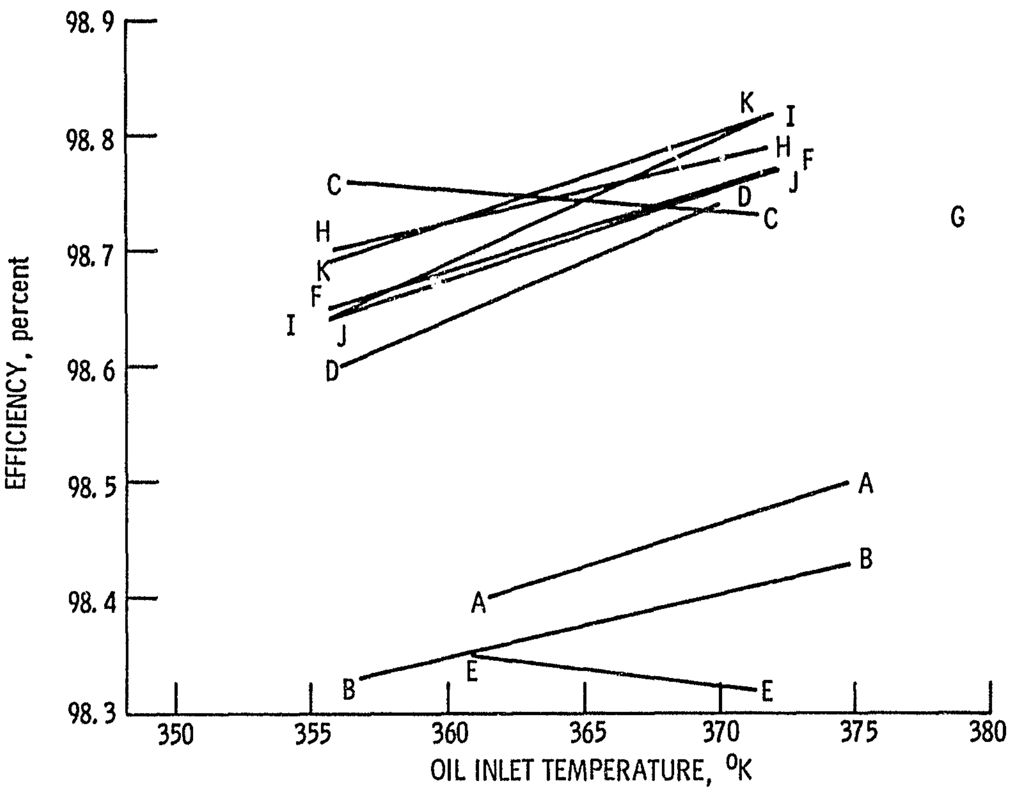

Figure 6. - Experimental efficiency correlated with inlet oil temperature.

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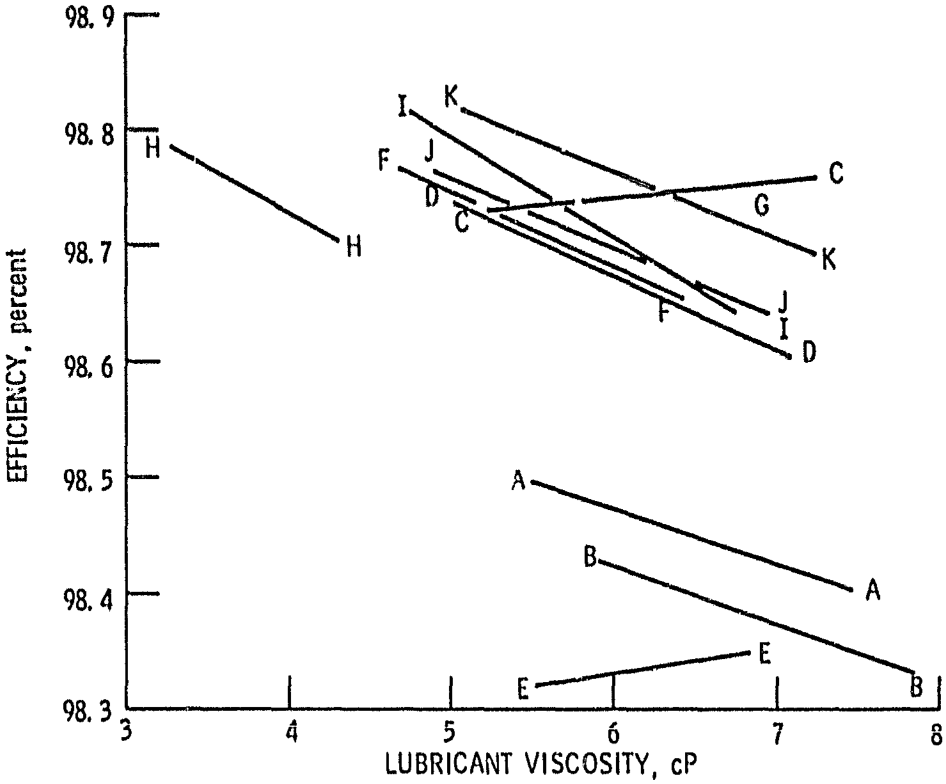

Figure 7. - Experimental efficiency correlated with lubricant viscosity.

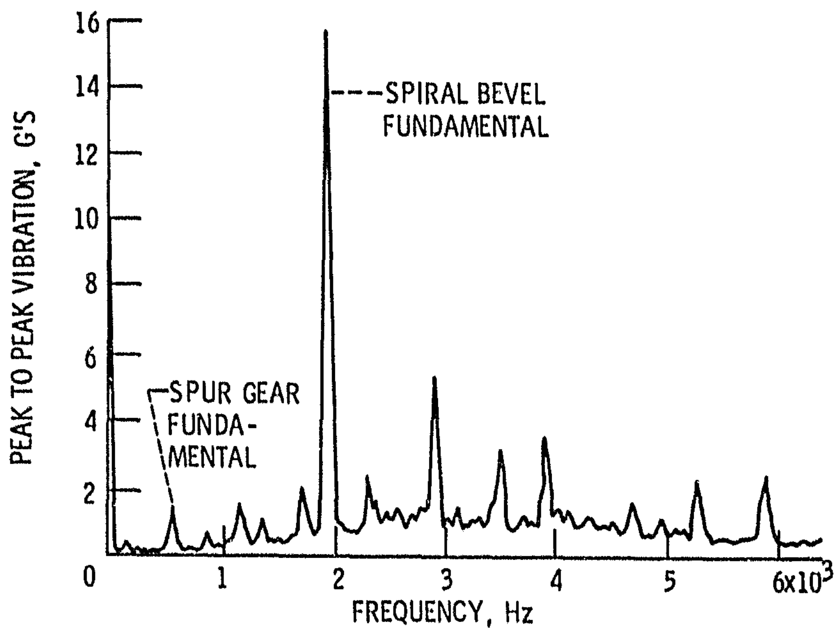

Figure 8. - Typical vibration spectrum.